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A Refrigerated Dynamic Seal

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A REFRIGERATED DYNAMIC SEAL

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ABSTRACT

A new sealing concept is proposed where the fluid to be sealed flows through a refrigerated housing or constriction, is frozen to the housing during the transient phase, and under steady state conditions provides a dynamic seal with essentially 'zero' leakage.

INTRODUCTION

In all applications the primary purpose of a seal is to prevent the loss of a fluid, i.e., minimize leakage. In turbomachine applications there are usually several seals between bearing supports. In many instances, improper design of these seals have been the cause of failures and derating of turbomachines, reference 1. Thus a secondary purpose for a seal, and often equally important as leakage control, is their use in prevention of dynamic instabilities.

In some applications one fluid may be sufficiently cooler than another and could serve as a heat sink for the fluid within the seal either by direct heat exchange, a heat pipe, or other such refrigeration devices. In these cases, the seal leakage and dynamic parameters as stiffness and dampening can be actively controlled by either the formation of a solid within the seal aperture or increasing the viscosity or both. The rotating member generates energy through viscous dissipation which is then balanced by the refrigeration load cited above.

In this paper a type of self sealing configuration is proposed for applications to turbomachines using conventional or cryogenic refrigeration technology to control the viscosity and solid plug buildup within the seal aperture.

ANALYSIS

Consider a shaft seal where either the working fluid can solidify or a solid plug can be formed through secondary fluid injection. Further consider a seal configuration such that the clearance to radius ratio $c/R = (b - t)/R \ll 1$, so that the problem can then be addressed as one-dimensional; refrigera-

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* a. While infinite at the critical point, viscosity has a minimum in the thermodynamic critical region.

b. At low temperatures (near the triple point) viscosity has a maximum for quantum fluids.

c. The temperature effect is usually strong for liquids ($\exp(a/T)$) and weak for gases, $T^{2/3}$. Further for a synthetic oil such as Mobile RL-714 Stock 509, a temperature change from 60° to 20° C (140° to 68° F), increases the viscosity by a factor of 4, reference 3.

d. One now must consider the stability of operating in the critical region with bearings etc. as η has minimum.

For gaseous nitrogen refrigerant flowing at 3 meters per second (10 ft/s) through a 6.4 millimeter (1/4 in.) diameter tube, the Reynolds number (Re) is 7800 and h_a is 28 watts per square meter (9 B/hr-ft^2-F). On the water side, for a leakage velocity of 1.8 meters per second (6 ft/s) with a hydraulic diameter of 2.5 millimeter (0.1 in.) the Reynolds number becomes 9700 and h_1 is 50 watts per square meter (16 B/hr-ft^2-F).

Using these values with b of 1.2 millimeter (0.05 in.) (i.e., t = 0), would require two minutes for an ice plug to form; for b of 0.75 millimeter (0.03 in.) would require less than one minute. The actual 'pinch off' clearance (b - t) is self limiting; for as soon as the clearance becomes very small, S becomes large and melts the ice. An additional feature is that the maximum temperature occurs within the aperture channel and not at the rotating shaft thereby preventing complete closure so long as the shaft is rotating.

EXPERIMENTAL RESULTS

An apparatus, shown in figure 3, was constructed to illustrate the principle of a self sealing system and to investigate its effectiveness for practical applications. The fluid to be sealed was water and nitrogen was the refrigerant. Apertures of the orifice and Borda types (ref. 4) were used as shaft seals and placed between bearing supports. With the shaft rotating, water was added to the reservior which leaked profusely, figure 4. As the nitrogen cooled an annular region around each seal the leakage stopped entirely, figure 5. The time required to stabilize the rotor and seal the system is generally shorter for the Borda type seal as would be expected due to better heat transfer and larger contact area. For eccentric seals or those rubbed or misaligned or badly damaged, the solid plug (ice layer) builds up quickly, and with sufficient refrigeration the leak stops completely.

For this system with 51 millimeters (2 in.) of water head at 12° C, the leak rate was 21 milliliters per second through the Borda aperture and 33 milliliters per second through the orifice aperature. At refrigerant loads greater than 200 watts, the time required to stop the leakage was less than 25 seconds for either aperture and so the simplified model appears conservative. The threshold refrigerant loads were found to be 80 and 170 watts for the Borda and orifice respectively.

Once the plug is formed and maintained, the seal should not 'wear' and should adjust to system perturbations.

POSSIBLE APPLICATIONS AND LIMITATIONS

1. The method can be applied to systems requiring very precise seal apertures. During the assembly and 'rub in' phase, a temperature gradient is imposed across some suitable seal material such that the interface is raised to a temperature $T_m + \Delta T$ where T_m is the melting temperature and $T_m > T_{0,max}$. Rotate the shaft system slowly and increase to operating speed; then bring the temperature back to equilibrium such that $T < T_m$ while checking torque and leakage. If not satisfactory, then repeat. Such a seal can be within microns of the rotating member and limited by the characteristic surface roughness (ref. 5).

2. A portable system with a flexible coolant system as a heat pipe or flex tube could be used as a quick fix to arrest leaking seals of operating systems until such time as it is shutdown for repairs. The concepts could

also be applied to a dual-spray system whereby a matrix material capable of 'holding' the refrigerant is sprayed onto a leak form one nozzle and the refrigerant from another.

3. For the dynamic applications, a rub will increase the amount of fluid within the passage. Continuity will require more fluid to be flow circumferentially and provide damping of the system. This aspect of enhanced stability can be as important as leakage control.

4. The maintenance of a thin fluid layer between the solid plug and the rotating member can significantly reduce the friction over that of dry sliding contact type seals.

5. It is possible to use the method in reverse, i.e., add heat to melt a region near the rotating member to facilitate less energy loss due to solid contact during rotation, an ablative method, reference 6.

6. This method can not be used as a static seal unless auxillary heat is available to melt a thin layer adjacent to the rotating member prior to startup.

7. The method can not be used to form a plug when the fluids are noncondensibles; however, the viscous effect on axial pressure drop can still be significant. Secondary fluid injection could be used to overcome this difficulty.

8. Due to the concentration effects of sudden ice formation, corrosion of the shaft material may occur.

SUMMARY

A new sealing concept is proposed where the fluid to be sealed flows through a refrigerated housing or constriction. As the fluid cools the viscosity increases and the resistance to leakage increases and in some cases leakage stops completely. In some cases this may be sufficient to establish the desired leakage rates and in others the temperature needs to be lowered below the freezing point of the fluid in the aperture to build and maintain a solid plug. Under steady operating conditions the viscous dissipation provided by the rotating member maintains a thin liquid film that can sustain large axial pressure gradients thereby providing a dynamic seal with essentially 'zero' leakage.

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6. Hendricks, R. C.: (NASA patent disclosure on uses of ablative materials in seals)

SYMBOLS

- b aperture static clearance
- c aperture dynamic clearance
- Cp specific heat
- d hydraulic diameter
- G mass flux
- H latent heat
- h heat transfer coefficient
- k thermal conductivity
- L ratio of heat transported by refrigerant to that conducted through the solid plug, eq. 5.
- N ratio of heat input form the seal to that absorbed by the refrigerant, eq. 4.
- Nu Nusselt number, hd/k
- Pr Prandtl number, Cp η/k
- R shaft radius
- Re Reynolds number, G d/η
- S viscous dissipation
- T temperature
- t time
- Δ ( ) increment
- ρ density
- Θ dimensionless time in terms of convection, conduction and latent energies at the interface, eq. 3.
- θ time
- η viscosity
- ω angular velocity

Subscripts

- a coolant side
- f fusion or solid
- i seal fluid side
- m melting
- max maximum
- o reference or stagnation

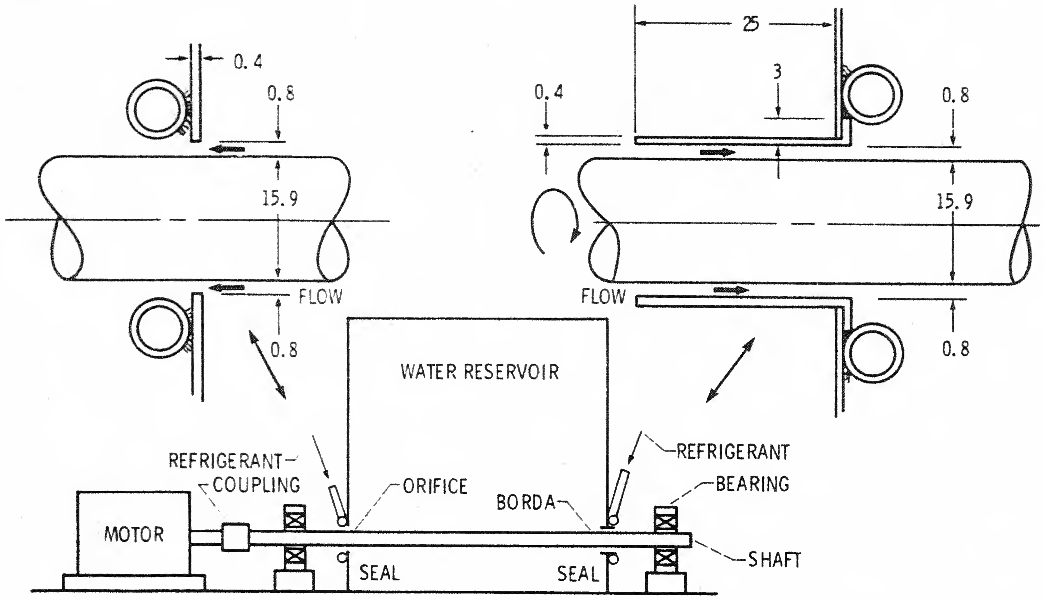

Figure 3. - Schematic of the experimental test apparatus for Borda and orifice type refrigerated seals. Dimensions in mm.

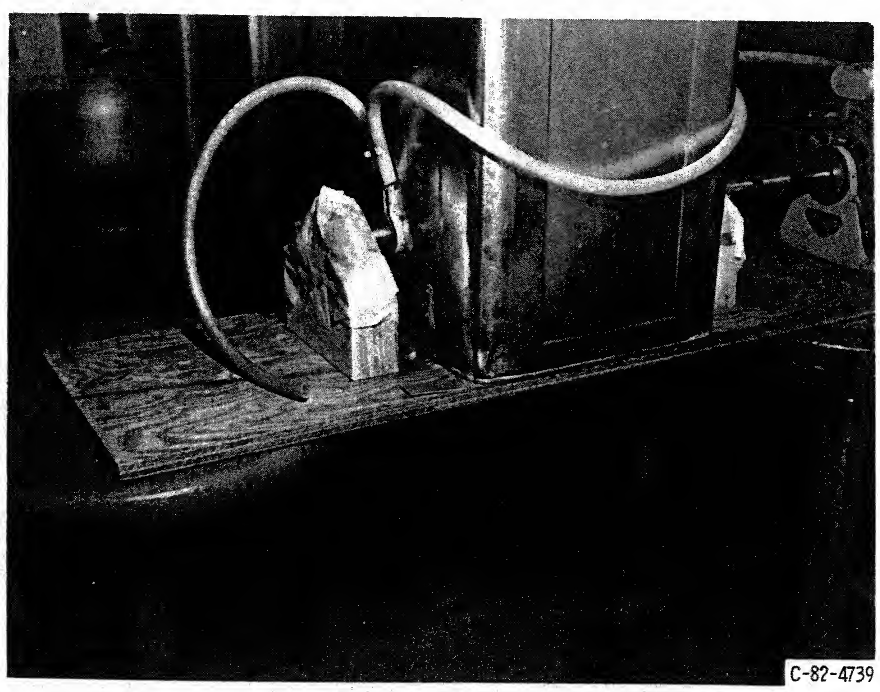

Figure 4. - Photograph of the non-refrigerated seal apparatus with rotation.

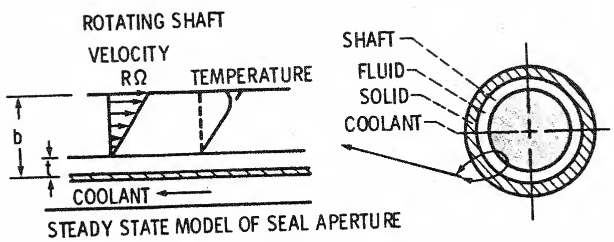

Figure 1. - Schematic of velocity and temperature profiles for a refrigerated seal configuration.

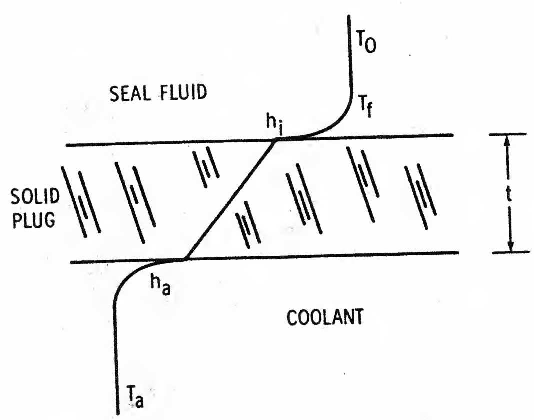

Figure 2. - Temperature profile for the refrigerated seal model.

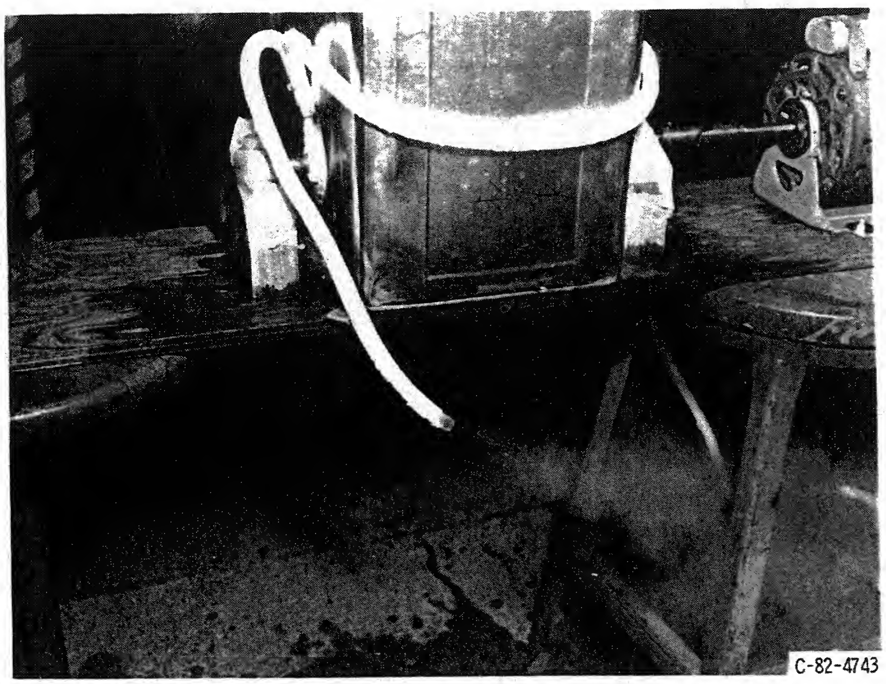

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Figure 5. - Photograph of the apparatus after application of refrigeration, with rotation.

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